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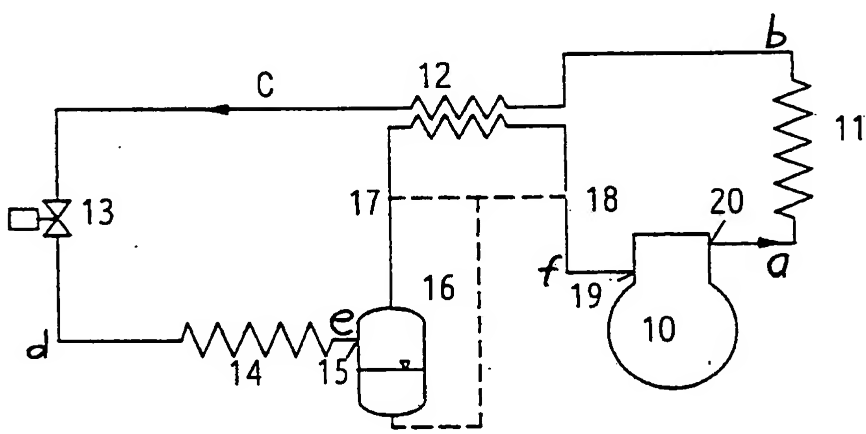

(57) Abstract

The present invention involves the regulation of specific enthalpy at evaporator inlet by deliberate use of the pressure and/or temperature before throttling for capacity control. Capacity is controlled by varying the refrigerant enthalpy difference in the evaporator, by changing the specific enthalpy of the refrigerant before throttling. In the super-critical state this can be done by varying the pressure and temperature independently. In a preferred embodiment this modulation of specific enthalpy is done by varying the pressure before throttling. The refrigerant is cooled down as far as it is feasible by means of the available cooling medium, and the pressure regulated to give the required enthalpy. Another embodiment involves modulation of enthalpy by variation of the refrigerant temperature before throttling. This is done by controlling the heat rejection from the device.

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Title of the invention

Trans-critical vapour compression cycle device.

Field of the invention

This invention relates to vapour compression cycle devices such as refrigerators, air-conditioning units and heat pumps, using a refrigerant operating in a closed circuit under trans-critical conditions, and more particularly, to methods for modulating and controlling the capacity of such devices.

Background of the invention

A conventional vapour compression cycle device for refrigeration, air-conditioning or heat pump purposes is shown in principle in Fig. 1. The device consists of a compressor (1), a condensing heat exchanger (2), a throttling valve (3) and a evaporating heat exchanger (4). These components are connected in a closed flow circuit, in which a refrigerant is circulated. The operating principle of a vapour compression cycle device is as follows: The pressure and temperature of the refrigerant vapour is increased by the compressor (1), before it enters the condenser (2) where it is cooled and condensed, giving off heat to a secondary coolant. The high-pressure liquid is then throttled to the evaporator pressure and temperature by means of the expansion valve (3). In the evaporator (4), the refrigerant boils and absorbs heat from its surroundings. The vapour at the evaporator outlet is drawn into the compressor, completing the cycle.

Conventional vapour compression cycle devices use refrigerants (as for instance R-12, CF_2Cl_2) operating entirely at sub-critical pressures. A number of different substances or

mixtures of substances may be used as a refrigerant. The choice of refrigerant is among others influenced by the condensation temperature, as the critical temperature of the fluid sets the upper limit for the condensation to occur. In order to maintain a reasonable efficiency, it is normally desirable to use a refrigerant with critical temperature at least 20-30K above the condensation temperature. Near-critical temperatures are normally avoided in design and operation of conventional systems.

The present technology is treated in full detail in the literature, e.g. the Handbooks of American Society of Heating, Refrigerating and Air Conditioning Engineers Inc., Fundamentals 1989 and Refrigeration 1986.

The ozone-depleting effect of todays common refrigerants (halocarbons) has resulted in strong international action to reduce or prohibit the use of these fluids. Consequently there is a urgent need for finding alternatives to the present technology.

Capacity control of the conventional vapour compression cycle device is achieved mainly by regulating the mass flow of refrigerant passing through the evaporator. This is done e.g. by controlling the compressor capacity, throttling or bypassing. These methods involve more complicated flow circuit and components, need for additional equipment and accesories, reduced part-load efficiency and other complications.

A common type of liquid regulation device is a thermostatic expansion valve, which is controlled by the superheat at the evaporator outlet. Proper valve operation under varying operating conditions is achieved by using a considerable part of the evaporator to superheat the refrigerant, resulting in a low heat transfer coefficient.

Furthermore, heat rejection in the condenser of the conventional vapour compression cycle takes place mainly at constant temperature. Therefore, thermodynamic losses occur due

to large temperature differences when giving off heat to a secondary coolant with large temperature increase, as in heat pump applications or when the available secondary coolant flow is small.

The operation of a vapour compression cycle under transcritical conditions has been formerly practiced to some extent. Up to the time when the halocarbons took over - 40-50 years ago - CO_2 was commonly used as a refrigerant, notably in ships refrigeration for provisions and cargo. The systems were designed to operate normally at sub-critical pressures, with evaporation and condensation. Occasionally, typically when a ship was passing tropical areas the cooling sea water temperature could be too high to effect normal condensation, and the plant would operate with supercritical conditions on the high-side. (Critical temperature for CO_2 ~31°C). In this situation it was practiced to increase the refrigerant charge on the high-side to a point where the pressure at the compressor discharge was raised to 90-100 bar, in order to maintain the cooling capacity at a reasonable level. CO_2 refrigeration technology is described in older literature, e.g. P. Ostertag "Kalteprozesse", Springer 1933 or H.J. MacIntire "Refrigeration Engineering", Wiley 1937.

The usual practice in older CO_2-systems was to add the necessary extra charge from separate storage cylinders. A receiver installed after the condenser in the normal way will not be able to provide the functions intended by the present invention.

Another possibility to increase the capacity and efficiency of a given vapour compression cycle device operating with supercritical high-side pressure is known from German patent 278095 (1912). This method involves two-stage compression with intercooling in the supercritical region. Compared to the standard system, this involves installation of an additional compressor or pump, and a heat exchanger.

The textbook "Principles of Refigeration" of W.B Gosney (Cambridge Univ. Press 1982) points at some of the peculiarities of near-critical pressure operation. It is suggested that increasing the refrigerant charge in the high-pressure side could be accomplished by temporarily shutting the expansion valve, so as to transfer some charge from the evaporator. But it is emphasized that this would leave the evaporator short of liquid, causing reduced capacity at the time when it is most wanted.

Objects of the invention

It is therefore an object of the present invention to provide a new, improved, simple and effective means for modulating and controlling the capacity of a trans-critical vapour compression cycle device, avoiding the above shortcomings and disadvantages of the prior art.

Another object of the present invention is to provide a vapour compression cycle avoiding use of CFC refrigerants, and at the same time offering possibility to apply several attractive refrigerants with respect to safety, environmental hazards and price.

Further object of the present invention is to provide a new method of capacity control, which involves operation at mainy constant refrigerant mass flow rate and simple capacity modulation by valve operation.

Still another object of the present invention is to provide a cycle rejecting heat at gliding temperature, reducing heat-exchange losses in applications where secondary coolant flow is small or when the secondary coolant is to be heated to a relatively high temperature.

Summary of the invention

The above and other objects of the present invention are achieved by providing a method operating normally at trans-critical conditions (i.e. super-critical high-side pressure, sub-critical low-side pressure) where the thermodynamic properties in the super-critical state are utilized to control the refrigerating and heating capacity of the device.

The present invention involves the regulation of specific enthalpy at evaporator inlet by deliberate use of the pressure and/or temperature before throttling for capacity control. Capacity is controlled by varying the refrigerant enthalpy difference in the evaporator, by changing the specific enthalpy of the refrigerant before throttling. In the super-critical state this can be done by varying the pressure and temperature independently. In a preferred embodiment this modulation of specific enthalpy is done by varying the pressure before throttling. The refrigerant is cooled down as far as it is feasible by means of the available cooling medium, and the pressure regulated to give the required enthalpy. Another embodiment involves modulation of enthalpy by variation of the refrigerant temperature before throttling. This is done by controlling the heat rejection from the device.

Brief description of the drawings

The invention will now be described in more detail, in the following referring to attatched drawings, Fig. 1, 2, 3, 4, 5, 6, 7 and 8, where:

Fig. 1 is a schematic representation of a conventional (sub-critical) vapour compression cycle device.

Fig. 2 is a schematic representation of a trans-critical vapour compression cycle device constructed in accordance

with a preferred embodiment of the invention. This embodiment includes a volume as an integral part of the evaporator system, holding refrigerant in the liquid state.

Fig. 3 is a schematic representation of a trans-critical vapour compression cycle device constructed in accordance with a second embodiment of the invention. This embodiment includes an intermediate pressure receiver connected directly into the flow circuit between two valves.

Fig. 4 is a schematic representation of a trans-critical vapour compression cycle device constructed in accordance with a third embodiment of the invention. This embodiment includes a special receiver to hold refrigerant as liquid or in the super-critical state.

Fig. 5 is a graph illustrating the relationship of pressure versus enthalpy of the trans-critical vapour compression cycle device of Fig. 2, 3 or 4, at different operating conditions.

Fig. 6 is a collection of graphs illustrating the control of refrigerating capacity by the method of pressure control in accordance with the present invention. The results shown are measured in a laboratory demonstration system built according to a preferred embodiment of the invention.

Fig. 7 is a collection of graphs illustrating the control of refrigerating capacity by control of the heat rejection, in accordance with the present invention. The results shown are measured in a laboratory demonstration system build according to a preferred embodiment of the present invention.

Fig. 8 is test results showing the relationship of temperature versus entropy of the trans-critical vapour compression cycle device of Fig. 2, operating at different high-side pressures, employing carbon dioxide as refrigerant

Detailed description of the invention

A trans-critical vapour compression cycle device according to the present invention includes a refrigerant, of which critical temperature is between the temperature of the heat inlet and the mean temperature of heat submittal, and a closed working fluid circuit where the refrigerant is circulated.

Suitable working fluids may be by the way of examples: ethylen (C_2H_4), diborane (B_2H_6), carbon dioxide (CO_2), ethane (C_2H_6) and nitrogen oxide (N_2O).

The closed working fluid circuit consists of a refrigerant flow loop with an integrated storage segment. Fig. 2 shows a preferred embodiment of the invention where the storage segment is an integral part of the evaporator system. The flow circuit includes a compressor 10 connected in series to a heat exchanger 11, a counterflow heat exchanger 12 and a throttling valve 13. The throttling valve can be replaced by an optional expansion device. An evaporating heat exchanger 14, a liquid separator/receiver 16 and the low-pressure side of the counterflow heat exchanger 12 are connected in flow communication intermediate the throttling valve 13 and the inlet 19 of the compressor 10. The liquid receiver 16 is connected to the evaporator outlet 15, and the gas phase outlet of the receiver 16 is connected to the counterflow heat exchanger 12.

The counterflow heat exchanger 12 is not absolutely necessary for the functioning of the device but improves its efficiency, in particular its rate of response to a capacity increase requirement. It also serves to return oil to the compressor. For this purpose a liquid phase line from the receiver (16) (shown with broken line in Fig. 2) is connected to the suction line either before the counterflow heat exchanger (12) at 17 or after it at 18, or anywhere between these points. The liquid flow, i.e. refrigerant and oil, is controlled by a suitable conventional liquid flow restricting device (not shown in the figure). By allowing some ex-

cess liquid refrigerant to enter the vapour line, a liquid surplus at the evaporator outlet is obtained.

In a second embodiment of the invention indicated in Fig. 3, the storage segment of the working fluid circuit includes a receiver 22 integrated in the flow circuit between a valve 21 and the throttling valve 13. The other components 10-14 of the flow circuit is identical to the components of the previous embodiment, although the heat exchanger 12 can be omitted without any great consequence. The pressure in the receiver 22 is kept intermediate the high-side and low-side pressures of the flow circuit.

In a third embodiment of the invention indicated in Fig. 4, the storage segment of the working fluid circuit includes a special receiver 25, where the pressure is kept between the high-side pressure and the low-side pressure of the flow circuit. The storage segment further consists of the valves 23 and 24 which are connected to the high pressure and low pressure part of the flow circuit respectively.

In operation, the refrigerant is compressed to a suitable supercritical pressure in the compressor 10, the compressor outlet 20 is shown as state "a" in Fig. 5. The refrigerant is circulated through the heat exchanger 11 where it is cooled to state "b", giving off heat to a suitable cooling agent, e.g. cooling air or water. If desired, the refrigerant can be further cooled to state "c" in the counterflow heat exchanger 12, before throttling to state "d". By the pressure reduction in the throttling valve 13, a two-phase gas/liquid mixture is formed, shown as state "d" in Fig. 3. The refrigerant absorbs heat in the evaporator 14 by evaporation of the liquid phase. From state "e" at the evaporator outlet, the refrigerant vapour can be superheated in the counterflow heat exchanger 12 to state "f" before it enters the compressor inlet 19, making the cycle complete. In the preferred embodiment of the invention, as shown in Fig. 2, the evaporator outlet condition "e" will be in the two-phase region due to the liquid surplus at the evaporator outlet.

Modulation of the trans-critical cycle device capacity is accomplished by varying the refrigerant state at the evaporator inlet, i.e. point "d" in Fig. 5. The refrigerating capacity per unit of refrigerant mass flow corresponds to the enthalpy difference between state "d" and state "e". This enthalpy difference is found as a horizontal distance in the enthalpy-pressure diagram, Fig. 5.

Throttling is a constant enthalpy process, thus the enthalpy in point "d" is equal to the enthalpy in point "c". In consequence, the refrigerating capacity (in kW) at constant refrigerant mass flow can be controlled by varying the enthalpy at point "c".

It should be noted that in the trans-critical cycle the high-pressure single-phase refrigerant vapour is not condensed but reduced in temperature in the heat exchanger 11. The terminal temperature of the refrigerant in the heat exchanger (point "b") will be some degrees above the entering cooling air or water temperature, if counterflow is used. The high-pressure vapour can then be cooled a few degrees lower, to point "c", in the counterflow heat exchanger 12. The result is, however, that at constant cooling air or water inlet temperature, the temperature at point "c" will be mainly constant, independent of the pressure level in the high side.

Therefore, modulation of device capacity is accomplished by varying the pressure in the highside, while the temperature in point "c" is mainly constant. The curvature of the isotherms near the critical point result in a variation of enthalpy with pressure, as shown in Fig. 5. The figure shows a reference cycle (a-b-c-d-e-f), a cycle with reduced capacity due to reduced high side pressure (a'-b'-c'-d'-e-f) and a cycle with increased capacity due to higher pressure in the high side (a"-b"-c"-d"-e-f). The evaporator pressure is assumed to be constant.

The pressure in the high-pressure side is independent of temperature, because it is filled with a single phase fluid.

To vary the pressure it is necessary to vary the mass of refrigerant in the high side, i.e. to add or remove some of the instant refrigerant charge in the high side. These variations must be taken up by a buffer, to avoid liquid overflow or drying up of the evaporator.

In the preferred embodiment of the invention indicated in Fig. 2, the refrigerant mass in the high side can be increased by temporarily reducing the opening of the throttling valve 13. Due to the incidentally reduced refrigerant flow to the evaporator, the excess liquid fraction at the evaporator outlet (15) will be reduced. The liquid refrigerant flow from the receiver 16 into the suction line is however constant. Consequently, the balance between the liquid flow entering and leaving the receiver 16 is shifted, resulting in a net reduction in receiver liquid content and a corresponding accumulation of refrigerant in the high pressure side of the flow circuit.

The increase in high side charge involves increasing pressure and thereby higher refrigerating capacity. This mass transfer from the low-pressure to the high-pressure side of the circuit will continue until balance between refrigerating capacity and load is found.

Opening of the throttling valve 13 will increase the excess liquid fraction at the evaporator outlet 15, because the evaporated amount of refrigerant is mainly constant. The difference between this liquid flow entering the receiver and the liquid flow from the receiver into the suction line, will accumulate. The result is a net transport of refrigerant charge from the high side to the low side of the flow circuit, with the reduction in the high side charge stored in liquid state in the receiver. By reducing the high-side charge and thereby pressure, the capacity of the device is reduced, until balance is found.

Some liquid transport from the receiver into the compressor suction line is also needed to avoid lubricant accumulation in the liquid phase of the receiver.

In the second embodiment of the invention indicated in Fig. 3, the refrigerant mass in the high side can be increased by simultaneously shutting the valve 21 and modulating the throttling valve 13 to provide the evaporator with sufficient liquid flow. This will reduce the refrigerant flow from the high side into the receiver through valve 21, while refrigerant mass is transferred from the low side to the high side by the compressor.

Reduction of high-side charge is obtained by opening the valve 21 while keeping the flow through the throttling valve 13 mainly constant. This will transfer mass from the high-side of the flow circuit to the receiver 22.

In a third embodiment of the invention indicated in Fig. 4, the refrigerant mass in the high side can be increased by opening the valve 24 and simultaneously reducing the flow through the throttling valve 13. By this, refrigerant charge is accumulated in the high-pressure side due to reduced flow through the throttling valve 13. Sufficient liquid flow to the evaporator is obtained by opening the valve 24.

A reduction in the high side charge can be accomplished by opening the valve 23 to transfer some refrigerant charge from the high side to the receiver. Capacity control of the device is thus accomplished by modulation of the valves 23 and 24, and simultaneously operating the throttling valve 13.

The preferred embodiment of the invention, as indicated in fig. 2 has the advantage of simplicity, with capacity control by operation of one valve only. Furthermore, the transcritical vapour compression cycle device built according to this embodiment has a certain self-regulating capability by adapting to changes in cooling load through changes in liquid content in the receiver 16, involving changes in highside charge and thus cooling capacity. In addition, the operation with liquid surplus at evaporator outlet gives favourable heat transfer characteristics.

The second embodiment, as indicated in Fig. 3, has the advantage of simplified valve operation. Valve 21 only regulates the pressure in the high side of the device, and the throttling valve 13 only assures that the evaporator is fed sufficiently. A conventional thermostatic valve can thus be used for throttling. Oil return to the compressor is easily achieved by allowing the refrigerant to flow through the receiver. This embodiment however does not offer the capacity control function at high-side pressures below the critical pressure. The volume of the receiver 22 must be relatively large since it is only operating between the discharge pressure and the liquid line pressure.

Still another embodiment as indicated in Fig. 4, has the advantage of operating as a conventional vapour compression cycle device, when it is running at stable conditions. The valves 23 and 24, connecting the receiver 25 to the flow circuit, are activated only during capacity control. This embodiment requires use of three different valves during periods of capacity change.

The latter embodiments has the disadvantage of higher pressure in the receiver, as compared to the preferred embodiment. The differences between the individual systems regarding design and operational characteristics are however not very significant.

Trans-critical vapour compression cycle devices built according to the described embodiments can be applied in several areas. The technology is well suitable in small and medium-sized stationary and mobile air-conditioning units, small and medium-sized refrigerators/freezers and in smaller heat pump units. One of the most promising applications is in automotive air-conditioning, where the present need for a new, non-CFC, lightweight and efficient alternative to R12-systems is urgent.

The above described embodiments of this invention are intended to be exemplative only and not limiting. It will be appreciated that it is also possible to control the capacity

of the trans-critical cycle device by keeping the high-side pressure mainly constant, and regulate the refrigerant temperature before throttling (state "c") by varying the circulation rate of cooling air or water. By reducing the flow of cooling fluid, i.e. air or water, the temperature before throttling will increase and the capacity will drop. Increased cooling fluid flow will reduce the temperature before throttling, and thereby increase the capacity of the device. Combinations of pressure and temperature control are also possible.

Examples

The practical use of the present invention for refrigeration or heat pump purposes is illustrated by the following examples, giving test results from a trans-critical vapour compression cycle device, built according to the embodiment of the invention shown in Fig. 2, employing carbon dioxide (CO_2) as refrigerant.

The laboratory test device uses water as heat source, i.e. the water is refrigerated by heat exchange with boiling CO_2 in the evaporator 14. Water is also used as cooling agent, being heated by CO_2 in the heat exchanger 11. The test device includes a 61 ccm reciprocating compressor (10) and a receiver (16) with total volume of 4 liters. The system also includes a counterflow heat exchanger (12) and liquid line connection from the receiver to point 17, as indicated in Fig. 2. The throttling valve 13 is operated manually.

Example 1

This example shows how control of refrigerating capacity is obtained by varying the position of the throttling valve 13, thereby varying the pressure in the high-side of the flow circuit. By variation of high-side pressure, the specific

refrigerant enthalpy at the evaporator inlet is controlled, resulting in modulation of refrigerating capacity at constant mass flow.

The water inlet temperature to the evaporator 14 is kept constant at 20°C, and the water inlet temperature to the heat exchanger 11 is kept constant at 35°C. Water circulation is constant both in the evaporator 14 and the heat exchanger 11. The compressor is running at constant speed.

Fig. 6 shows the variation of refrigerating capacity (Q), compressor shaft work (W), highside pressure (p_H), CO_2 mass flow (m), CO_2 temperature at evaporator outlet (t_e), CO_2 temperature at the outlet of heat exchanger 11 (t_b) and liquid level in the receiver (h) when the throttling valve 13 is operated as indicated at the top of the figure. The adjustment of throttling valve position is the only manipulation.

As shown in the figure, capacity (Q) is easily controlled by operating the throttling valve (13). It is further clear from the figure that at stable conditions, the circulating mass flow of CO_2 (m) is mainly constant and independent of the cooling capacity. The CO_2 temperature at the outlet of heat exchanger 11 (t_b) is also mainly constant. The graphs show that the variation of capacity is a result of varying high side pressure (p_H) only.

It can also be seen from the diagram that increased highside pressure involves a reduction in the receiver liquid level (h), due to the CO_2 charge transfer to the highpressure side of the circuit.

Finally, it can be noted that the transient period during capacity increase is not involving any significant superheating at the evaporator outlet, i.e. only small fluctuations in t_e.

Example 2

With higher water inlet temperature to heat exchanger 11 (e.g. higher ambient temperature), it is necessary to increase the high side pressure to maintain a constant refrigerating capacity. Table 1 shows results from tests run at different water inlet temperature to heat exchanger 11 (t_w).

The water inlet temperature to the evaporator is kept constant at 20°C, and the compressor is running at constant speed.

As the table shows, the cooling capacity can be kept mainly constant when the ambient temperature is rising, by increasing the high side pressure. The refrigerant mass flow is mainly constant, as shown. Increased high-side pressures involve a reduction in receiver liquid content, as indicated by the liquid level readings.

Table 1

Inlet temperature (t_w)	35.1	45.9	57.3	°C
Refrigerating capacity (Q)	2.4	2.2	2.2	kW
High side pressure (p_H)	84.9	94.3	114.1	bar
Mass flow (m)	0.026	0.024	0.020	kg/s
Liquid level (h)	171	166	115	mm

Example 3

This example illustrates the possibility to modulate and control the capacity of the device by adjustment of the flow of coolant (e.g. air or water) circulating through heat exchanger 11, keeping the high-side pressure constant.

Fig. 7 shows the variation of refrigerating capacity (Q) when the circulation rate of cooling water (m_w) is regulated as shown at the top of the figure. The mass flow of CO_2 (m), the high-side pressure (p_H) and the water inlet temperature to heat exchanger 11 (t_i) are kept constant. The compressor is running at constant speed and both the temperature and flow rate of water entering the evaporator are kept constant.

The refrigerating capacity is easily controlled by variation of the water flow, as shown in the figure. Mass flow of CO_2 is mainly constant.

Example 4

Fig. 8 is a graphic representation of trans critical cycles in the entropy/temperature diagram. The cycles shown in the diagram are based on measurements on the laboratory test device, during operation at five different high-side pressures. The evaporator pressure is kept constant. refrigerant is CO_2.

The diagram gives a good impression of the capacity control principle, indicating the changes in specific enthalpy (h) at evaporator inlet caused by variation of the high-side pressure (p).

Claims

1. Method for regulation of heating/cooling capacity of a vapour compression cycle device comprising a compressor (10), a heat exchanger (11), a throttling means (13) and a evaporating heat exchanger (14) connected in series forming an integral closed circuit applying a refrigerant operating under trans-critical conditions, characterized in that the capacity is regulated and controlled by variation of specific enthalpy of the supercritically pressurized refrigerant at the inlet of the throttling means (13).

2. The method according to claim 1, characterized in that the capacity regulation is performed by variating the supercritical refrigerant pressure at the inlet of the throttling means (13), by variation of the instant refrigerant charge in the high pressure side of the circuit.

3. The method according to claim 1, characterized in that the capacity regulation is obtained by variation of the refrigerant temperature at the inlet of the throttling means (13) by controlling the flow rate of the heat exchanging medium absorbing heat in the heat exchanger (11).

4. The method according to claim 2, characterized in that the throttling means (13) is applied as steering means to variate the liquid refrigerant inventory of a receiver (20) connected between the evaporator (14) and the compressor (10) in the low pressure side of the circuit and that a heat exchanger (12) is included between the receiver (20) and the compressor (10) to exchange heat from the high pressure gas for the purpose of evaporating liquid supplied from the receiver (20) in order to rapidly increase the charge build-up in the high pressure side without dry-up of the evaporator (14) and at the same time return oil to the compressor (10).

6. The method according to claim 2, c h a r a c t e r i z e d i n t h a t variation of instant refrigerant charge in the high pressure side of the flow circuit is obtained by modulating the valve (21) and the throttling means (13) to vary the supercritically pressurized refrigerant charge in a receiver (22) installed in the flow circuit between the valve (21) and the throttling means (13).

7. The method according to claim 2, c h a r a c t e r i z e d i n t h a t variation of instant refrigerant charge in the high pressure side of the flow circuit is obtained by continuously regulating the removal or filling of refrigerant to or from a storage device (25) connected to the high and low pressure sides of the flow circuit by means of pipes with valves (23, 24) and keeping the pressure in the storage device (25) intermediate the high side and the low side pressures.

8. The method according to one or more preceding claims, c h a r a c t e r i z e d i n t h a t the refrigerant is carbon dioxide.

9. The method according to one or more preceding claims, c h a r a c t e r i z e d i n t h a t the trans-critical vapour compression cycle device is applied in automotive air-conditioning.

AMENDED CLAIMS

[received by the International Bureau on 30 April 1990 (30.04.90); original claims 1-9 replaced by amended claims 1-9 (3 pages)]

1. A method for regulating the capacity of a vapor compression cycle comprising a compressor (10), a cooler (11), throttling means (13) and an evaporator (14) connected in series forming an integral closed circuit operating at supercritical pressure on the high pressure side of the cycle,
c h a r a c t e r i z e d i n t h a t
the capacity regulation is achieved by variation of the instant refrigerant charge in the high pressure side of the circuit.

2. Method according to claim 1,
c h a r a c t e r i z e d i n t h a t
the capacity regulation is based on modulation of the supercritical pressure and conducted by varying the liquid inventory of a low pressure refrigerant receiver (16) situated intermediate the evaporator (14) and the compressor (10) applying solely throttling means (13) as capacity steering means.

3. Method according to claim 1,
c h a r a c t e r i z e d i n t h a t
variation of the instant refrigerant charge in the high pressure side of the flow circuit is obtained by modulating the valve (21) and the throttling means (13) to vary the supercritically pressurized refrigerant charge in a receiver (22) installed in the flow circuit between the valve (21) and the throttling means (13).

4. Method according to claim 1,
c h a r a c t e r i z e d i n t h a t
variation of the instant refrigerant charge in the high pressure side of the flow circuit is obtained by continuously regulating the removal or filling of refrigerant to or from a storage device (25)

connected to the high and low pressure sides of the flow circuit by means of pipes with valves (23,24) and keeping the pressure in the storage device (25) intermediate the high side and the low side pressures.

5. Method according to claim 2, 3 or 4,
c h a r a c t e r i z e d i n t h a t
the evaporator outlet condition is maintained as a two phase mixture of vapor and liquid providing a liquid surplus at the low pressure inlet of an additional heat exchanger (12) where the low pressure refrigerant is subjected to evaporation and superheating prior to inlet to the compressor by heat from the high pressure refrigerant.

6. Method according to one or more preceding claims,
c h a r a c t e r i z e d i n t h a t
the refrigerant is carbon dioxide.

7. An automotive air conditioning device comprising a compressor (10), a cooler (11), throttling means (13) and an evaporator (14) connected in series forming an integral closed circuit,
c h a r a c t e r i z e d i n t h a t
the refrigerant is compressed to a supercritical pressure on the high pressure side of the circuit, and where the throttling means (13) are applied to modulate the capacity of the device by varying the liquid inventory of a low pressure receiver (16) situated intermediate the evaporator (14) and the compressor (10) causing variation in the supercritical high side pressure.

8. Device according to claim 7, characterized in that a heat exchanger (12) is additionally provided having a low pressure inlet (17) in communication with the receiver (16) and a high pressure inlet communicating with the outlet of the cooler (11), the heat exchanger being situated in the circuit intermediate the receiver (16) and the compressor (10).

9. Device according to claim 7 or 8, characterized in that the refrigerant is carbon dioxide.

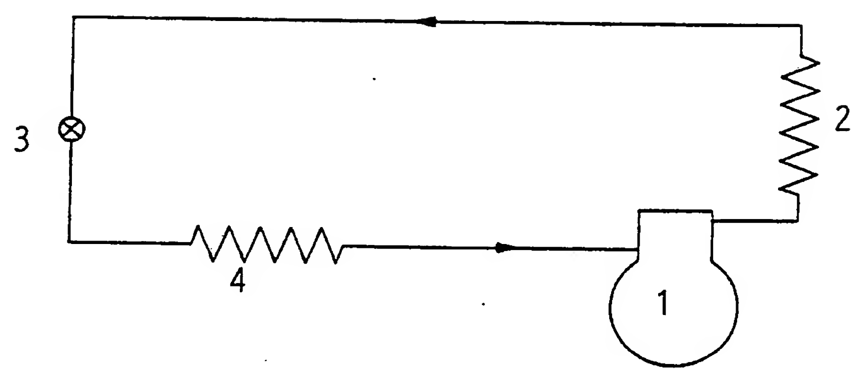


Fig. 1

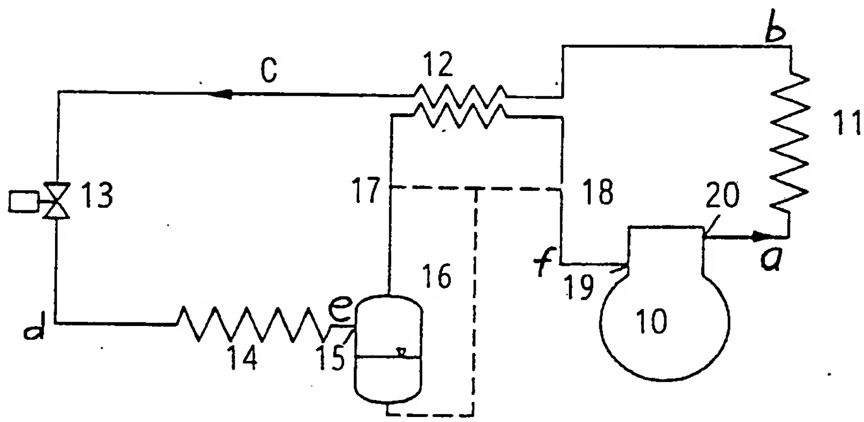


Fig. 2

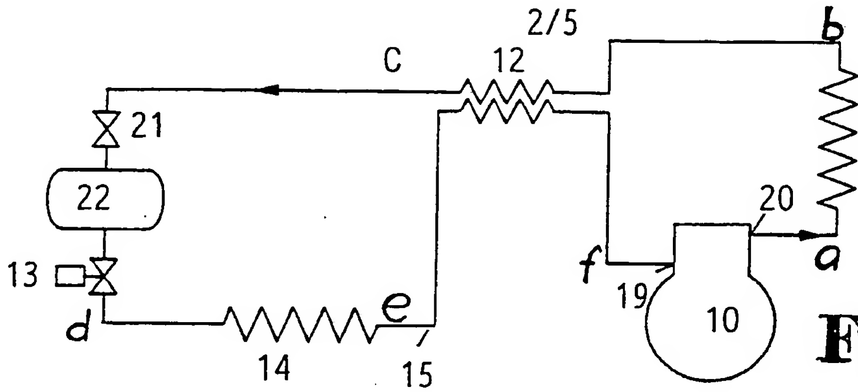


Fig. 3

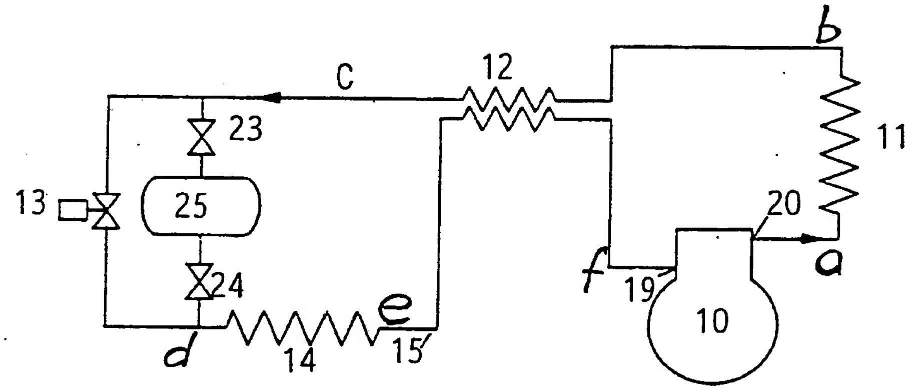


Fig. 4

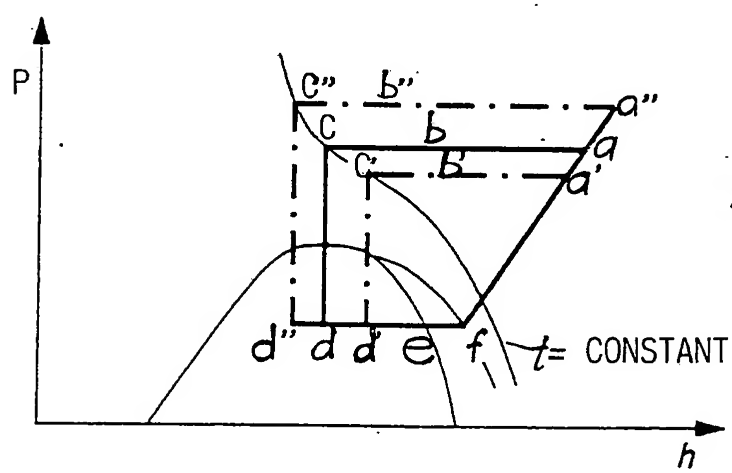


Fig. 5

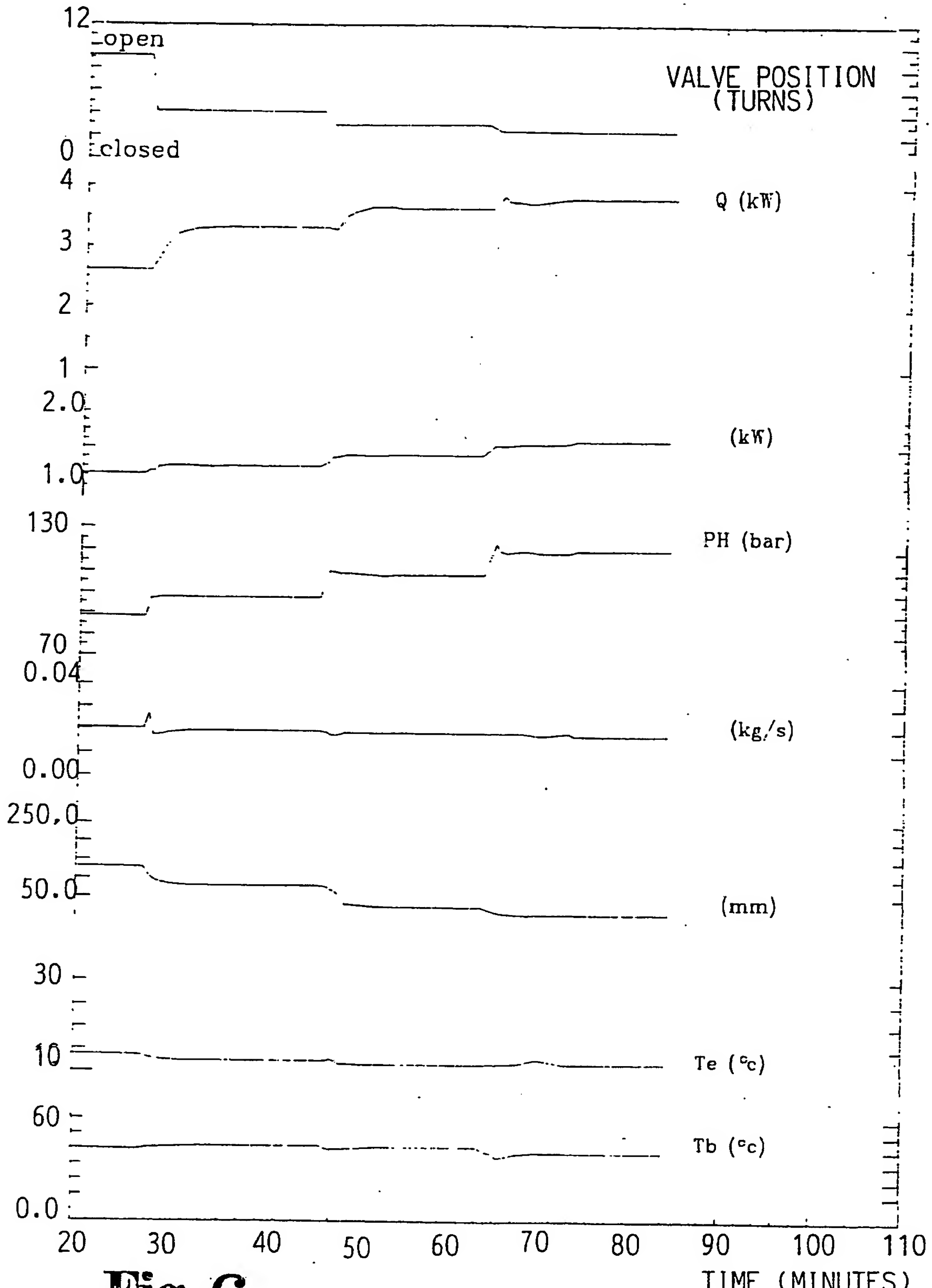

12
open
VALVE POSITION (TURNS)
0
closed
4
3
2
1
Q (kW)
2.0
1.0
(kW)
130
70
PH (bar)
0.04
0.00
(kg/s)
250.0
50.0
(mm)
30
10
Te (°c)
60
0.0
Tb (°c)
20
30
40
50
60
70
80
90
100
110
TIME (MINUTES)

Fig. 6

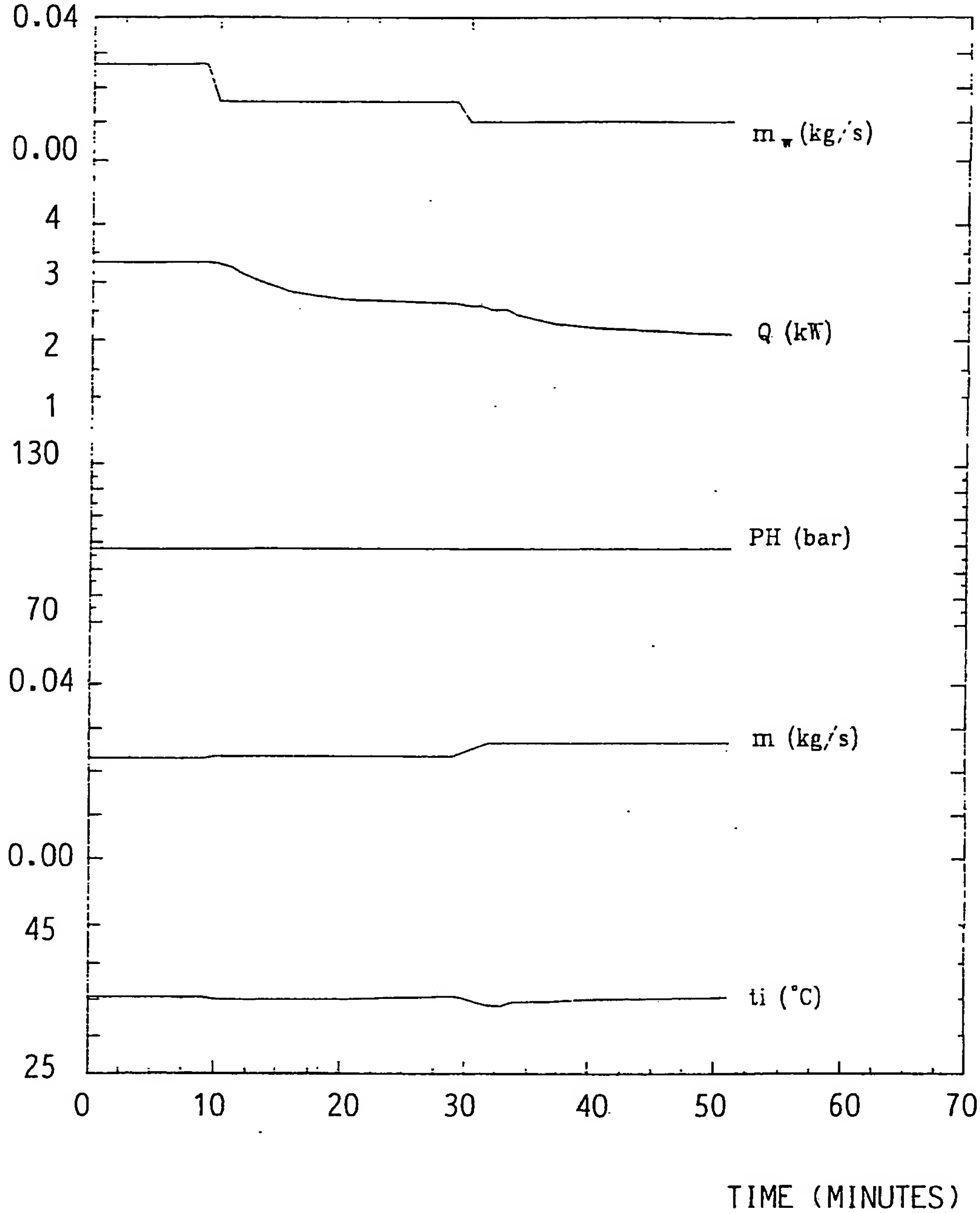
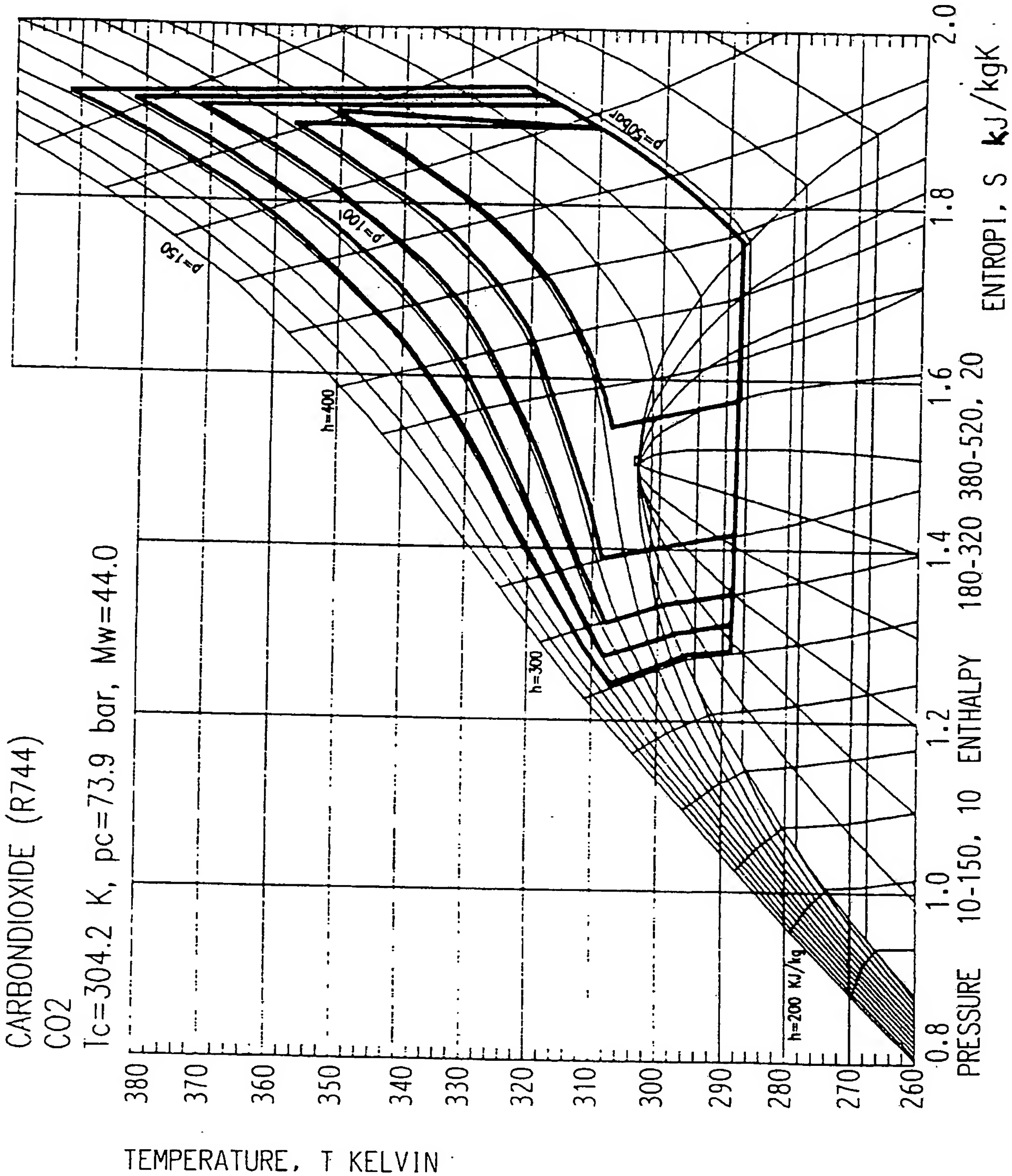

0.04
0.00
m_w (kg/s)
4
3
2
1
Q (kW)
130
PH (bar)
70
0.04
m (kg/s)
0.00
45
ti (°C)
25
0
10
20
30
40
50
60
70
TIME (MINUTES)

Fig.7


CARBONDIOXIDE (R744)
CO2
Tc=304.2 K, pc=73.9 bar, Mw=44.0
TEMPERATURE, T KELVIN
380
370
360
350
340
330
320
310
300
290
280
270
260
PRESSURE 10-150, 10 ENTHALPY 180-320 380-520, 20 ENTROPI, S kJ/kgK
0.8
1.0
1.2
1.4
1.6
1.8
2.0
p=50bar
p=100
p=150
h=400
h=300
h=200 KJ/kg

INTERNATIONAL SEARCH REPORT

International Application No PCT/NO 89/00089

I. CLASSIFICATION OF SUBJECT MATTER (if several classification symbols apply, indicate all) [6]

According to International Patent Classification (IPC) or to both National Classification and IPC

IPC4: F 25 B 1/00

II. FIELDS SEARCHED

Minimum Documentation Searched [7]

Classification System	Classification Symbols
IPC4	F 25 B

Documentation Searched other than Minimum Documentation to the Extent that such Documents are Included in the Fields Searched [8]

SE,DK,FI,NO classes as above

III. DOCUMENTS CONSIDERED TO BE RELEVANT [9]

Category [*]	Citation of Document, [11] with indication, where appropriate, of the relevant passages [12]	Relevant to Claim No. [13]
Y	US, A, 4205532 (BRENAN) 3 June 1980, see page 3, line 30 - page 4; page 7, line 1 - line 15; figure 1 --	1-4,8
Y	US, A, 1408453 (J.C. GOOSMANN) 7 March 1922, see page 4 - page 5; figure 1 --	1-4,8
A	US, A, 3872682 (SHOOK) 25 March 1975, see the whole document --	1
A	US, A, 4679403 (YOSHIDA ET AL) 14 July 1987, see the whole document --	1
A	US, A, 3400555 (E.G.U. GRANRYD) 10 September 1968, see the whole document --	1-8

* Special categories of cited documents: [10]

"A" document defining the general state of the art which is not considered to be of particular relevance

"E" earlier document but published on or after the international filing date

"L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)

"O" document referring to an oral disclosure, use, exhibition or other means

"P" document published prior to the international filing date but later than the priority date claimed

"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step

"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art.

"&" document member of the same patent family

IV. CERTIFICATION

Date of the Actual Completion of the International Search	Date of Mailing of this International Search Report
8th December 1989	1989 -12- 14
International Searching Authority	**Signature of Authorized Officer**
SWEDISH PATENT OFFICE	Inger Löfving

Form PCT/ISA/210 (second sheet) (January 1985)

International Application No. PCT/NO 89/00089

III. DOCUMENTS CONSIDERED TO BE RELEVANT (CONTINUED FROM THE SECOND SHEET)		
Category *	Citation of Document, with indication, where appropriate, of the relevant passages	Relevant to Claim No
A	DE, C, 278095 (RUDOLPH PLANK) 19 September 1914, see the whole document --	1-8
A	US, A, 3844131 (GIANNI ET AL) 29 October 1974, see the whole document --	1-8

ANNEX TO THE INTERNATIONAL SEARCH REPORT
ON INTERNATIONAL PATENT APPLICATION NO. PCT/NO 89/00089

This annex lists the patent family members relating to the patent documents cited in the above-mentioned international search report.

Patent document cited in search report	Publication date	Patent family member(s)	Publication date
US-A- 4205532	03/06/80	DE-A- 2819276 GB-A- 1544804	09/11/78 25/04/79
US-A- 1408453	07/03/22	NONE	
US-A- 3872682	25/03/75	GB-A- 1458945 CA-A- 1005247	15/12/76 15/02/77
US-A- 4679403	14/07/87	EP-A- 0174027 JP-A- 61066053 GB-A-B- 2163244 JP-A- 61262549	12/03/86 04/04/86 19/02/86 20/11/86
US-A- 3400555	10/09/68	NONE	
DE-C- 278095	19/09/14	NONE	
US-A- 3844131	29/10/74	NONE	